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(54) **TRANSMISSION OF THE TOROIDAL-RACE ROLLING-TRACTION TYPE.**

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Description

This invention relates to continuously-variable-ratio transmissions (which will be referred to as CVT's) of the toroidal-race, rolling-traction type and in particular to a roller control system for a CVT of the toroidal race, rolling fraction type as defined in the preamble of claim 1 and known from GB-A-1 600 972. It relates in particular to the variators, that is to say the ratio-varying units, of such transmissions in which rollers of variable orientation transmit traction between coaxial and part-toroidal input and output grooves or races, formed on coaxial and rotatable input and output discs respectively. By simultaneously altering the radius from the common axis of the discs at which the rollers make rolling contact with the two races, the relative speeds of the two discs change, resulting ,in a change in the transmitted ratio. While the prior art teaches and the invention will be described with relation to toruses of circular cross-section, the invention includes CVT's in which the torus is generated by rotating any closed figure, of generally circular outline, about a generator line.

Patent applications in this art, relating especially to automobile transmissions, have been filed regularly from at least the 1920's onwards, see US-A-1865102 as an example. In that specification, as in many others in this art, there are two input races and two output races and a set of three rollers transmits drive from each input race to its corresponding output race, all rollers in the variator being constrained at all times to contact their respective input race at a common first radius and their respective output race at a common second radius.

It has been common practice in the art to mount each roller so that it spins about an axle mounted in a supporting member usually called a carriage, and to connect the carriages of all the rollers in one set so that those carriages move in unison when required so as to change the transmitted ratio, and between such movements to hold their associated rollers steady so that they all transmit the same ratio in the manner already described.

In the accompanying drawings, Figures 1 to 3 all show the same, known type of variator. They are also all simplified and generally diagrammatic, and should be studied together because certain parts shown in one of them are omitted from one or both of the others. Figure 1 is an axial section through the variator, Figure 2 shows the roller-supporting mechanism in a section on the line II-II in Figure 1, and Figure 3 is a section on the line III-III in Figure 2. As shown in Figure 1, an input shaft 1 is rotatable about an axis 2, is driven by a prime mover 3 and carries two input discs 4 and 5 formed with part-toroidal races 6 and 7 respectively. Disc 5 is fixed to shaft 1, while a keyed connection 8 prevents mutual rotation between the shaft and disc 4 but allows limited relative axial movement. Disc 4 acts as a piston within a cylindrical cap 9 which is fixed to shaft 1, and the chamber 10 within the cap is connected to a pressurized fluid source 11. A single output disc 13, formed on its opposite faces with part-toroidal races 14 and 15, is mounted in a bearing 16 with freedom to rotate about input shaft 1 and to make limited relative movement axially. Disc 13 constitutes the output member of the variator and a gear 17, formed on the rim of the disc, engages with the final drive of the transmission (not shown) by way of a gear 18 rotatable on a support fixed relative to the variator casing 19. Race 14 conforms to the surface of the same torus as race 6, and races 15 and 7 are similarly related. A set of three rollers 20, which are equi-spaced around axis 2 but of which only one is shown, make rolling contact with races 6 and 14 and so transmit drive from input disc 4 to output disc 13. Rollers 20 are mounted in a supporting frame 21. A second and symmetrically-arranged set of rollers 25, mounted on a supporting frame 26, transmit drive from race 7 to race 15 formed on the opposite face of output disc 13. The necessary hydraulic end load to urge the discs and rollers firmly into contact with each other by way of an intervening thin film of fluid, so that they transmit the required driving power to the final drive by way of gear 18 in a manner well known in the art, is generated by the fluid in chamber 10. As already stated, input disc 4 and output disc 13 can make slight axial movements in response to that load.

The two roller supporting frames 21 and 26 are essentially similar, and frame 21 is illustrated best in Figure 2. It comprises a frame member 30 of generally triangular shape having a central aperture 31 to accommodate the shaft 1. Each roller 20 spins about an axis 33 on an axle 32 mounted in a carriage 34 which encompasses the roller along a line 35 passing through the roller centre 22 but leaves the two roller segments that are furthest from that line unobstructed, so that the roller can contact the races 6, 14 as already described. To change the transmitted ratio, each roller and its associated carriage 34 must be able to pivot about the same line 35 with which the carriage 34 is itself aligned, and one of the means well known in the art for inducing such pivotal movement is to impose "tangential shift" - that is to say a movement generally tangential to the centre circle of the common torus of races 6 and 14 - upon the roller and carriage. In Figures 2 and 3, which illustrate a mechanism that is known generally in the art and is particularly similar to what is described in patent specification GB-A-1395319, both the tangential shift and the resulting pivotal movement are facilitated by mounting ball ends 37, 38 at opposite extremities of carriage 34, the two ball centres both lying on line 35. End 37 slides within a cylindrical socket 39 mounted on frame 30, while end 38 is captive within a ball-shaped socket formed in a piston 40, which slides within a cylinder 41 also mounted on frame 30. The chamber 42 of cy-

linder 41 is connected by way of control valve means 43 to the same pressurised fluid source 11 by which the end load chamber 10 is supplied. By using valve 43 to vary the fluid pressure within chamber 42, piston 40 imparts tangential shift to carriage 34. As already referred to, those movements will have the effect of causing the carriage and its roller 20 to tilt about line 35, and so to change the transmitted ratio.

The centre 22 of each roller 20 must at all times lie on the centre circle of the imaginary torus to whose surfaces races 6 and 14 conform, and when the ratio unit is in equilibrium - that is to say, when the transmitted ratio is constant for the time being - the spin axis 33 of each roller intersects the variator axis 2. In order that ratio-change should be brought about by a combination of components of tangential shift and rotation, as just described, a further geometrical feature is desirable and is illustrated in Figure 3. This feature is that while the roller centre 22 lies in the central plane 50 of the imaginary torus at all times, ball end 37 lies to one side of that plane and ball end 38 to the other side, so that line 35 is inclined to plane 50 at an angle C known in the art as the castor angle. The effect of this angle may be explained as follows. If discs 4, 13 are rotating as indicated by arrows 51 and 52, the transmission of torque by rollers 20 between races 6 and 14 produces a torque reaction on each roller carriage 34, urging the associated piston 40 into its cylinder 41. For the transmission to be in equilibrium, two conditions must be fulfilled. Firstly the spin axis 33 of each roller must intersect the variator axis 2. Secondly the force exerted upon piston 40 by the fluid in cylinder 41 must be equal and opposite to the force which the torque reaction exerts upon the roller carriage, both forces being measured in a plane at right angles to the variator axis 2. If now the fluid pressure is increased in cylinder 41, driving the piston 40 downwards (as shown in Figure 3) against the direction of the discs/roller torque reaction, equilibrium is destroyed because the cylinder and torque reaction forces are no longer in balance. The roller axis 33 will therefore no longer intersect the variator axis 2. As a result a steering force is imposed on the roller by the discs 4 and 13 so as to tilt the carriage 34 about line 35, until equilibrium is restored when the cylinder and torque reaction forces are in balance again, and when axis 33 once more intersects axis 2, the degree of tilt (which is proportional to the resulting change in transmitted ratio) being dependent upon the size of the initial tangential displacement or shift, and of the castor angle. Tangential shift in the opposite direction, which in the known variator shown in Figure 3 will be brought about by a reduction in fluid pressure in cylinder 41, will result in the roller tilting in the opposite direction.

A fundamental feature of variators of the type just described in outline, and described in greater detail in patent specification GB-A-1395319 for example, is that they are of "force-balance" type. That is to say, one of the conditions that must be fulfilled for equilibrium of the transmission at any given ratio value is that the torque reaction force and the hydraulic force acting upon the carriage piston must be in balance. If either of these forces changes, equilibrium is lost until the forces .,are brought into balance once more. This feature distinguishes transmissions as shown in GB-A-1395319, and transmissions according to the present invention also, from an older generation of CVT's of the toroidal-race, rolling-traction type in which roller and carriage are positioned by mechanical means which are not themselves responsive to the torque reaction forces to which the rollers and carriages, once positioned, are themselves subjected. Patent specification US-A-2130314 describes a mechanical positioning system of this kind, in which one end of the roller carriage is connected by a ball-and-socket joint to a control pinion. The transmitted ratio is varied by turning the pinion, so changing the orientation of the carriage by altering the location of its point of connection to the pinion. However, the carriage/pinion connection is such that the torque reaction experienced at the disc/roller interfaces through the carriage acts upon the pinion in a direction substantially parallel to its axis of rotation. No useful balance between the torque reaction force and the force applied to the pinion to turn it is therefore possible, and means other than force balance must therefore be found to ensure that the pinion always seeks the rotary position at which the roller transmits the ratio required by the instantaneous prevailing conditions.

In the known mechanism of force-balance type shown in Figures 2 and 3 the ball ends 37, 38 can move axially and simultaneously within their respective cylinders so that the line 35 moves bodily, and each carriage 34 can rotate about the instantaneous position of line 35. However, because the carriage is located at both ends, it has no freedom to rotate about any other axis. Figure 4 of the drawings of GB-A-1600972 (equivalent to US-A-4281559) shows in accordance with the features of the preamble of claim 1 another variety of known mechanism in which the roller carriage (83) is fast with the head (82) of the single piston by which the position of the roller (13) is controlled. As with the two known mechanisms just described, this carriage is capable of translational movement along a line (the axis of movement of piston head (82)) and of rotation about that line, but has no freedom to rotate about any other axis. It should also be noted that in the CVT shown in Figure 4 of GB-A-1600972 the two rotors (10, 12) between which the roller (13) transmits traction must themselves be capable of simultaneous and equal movements, in a direction parallel to the main axis of the CVT, to accommodate displacements of the roller (13) by piston (82); the requirement for such movement of the rotors naturally introduces further complexity and expense for the CVT as a whole.

Another known design of CVT of the force-balance type is shown and described in US-A-3933054, in which the traction forces experienced by each roller ((40 - 42) in the drawings) are balanced by the hydraulic force acting on a piston (66). As to how to synchronise this balance of forces with the desired value of the transmitted ratio, the teaching of US-A-3933054 is clear. Each roller carriage is connected by a hinged joint (pin 47) to the mechanism on which the piston (66) is mounted. The carriage also carries a cam follower (50) which engages with a cam slot (70) secured to the transmission casing. As already explained, in the description of Figures 1 to 3, two conditions must be fulfilled if a transmission of this type is to be in equilibrium. Therefore when the transmission of US-A-3933054 falls out of equilibrium, two related but distinct motions must take place in order to restore it. Firstly there is a generally axial movement of each piston (66) within its cylinder (65), until a new torque reaction at the disc/roller interfaces balances a new hydraulic force exerted upon the piston by the fluid within the cylinder. Second, the angle of tilt of the roller (40 - 42) must change until the roller axis once more intersects the drive axis (D) of the transmission. US-A-3933054 teaches that the slot/follower (70/50) engagement is essential to achieve this second motion. In response to the first motion of the piston, the follower (50) is forced to move along the slot (70) so changing the angle of tilt of the roller (40 - 42) and thus the transmitted ratio. This requires both pivoting at the hinged joint (47), and rotation of the piston 66 about its axis within its cylinder (65). Now the axis of the hinged joint (47) intersects the line joining the two points of contact of the roller (41) with the disc grooves (30, 31), so without the engagement of follower (50) and slot (70) as taught by US-A-3933054, the tilt angle of each roller (40 - 42) in response to any loss of equilibrium of the transmission, would be indeterminate. As taught by US-A-3933054, therefore, for effective operation each assembly of carriage and roller thus requires four points of contact with adjacent mechanism, namely the contact between the roller and the two grooves between which it is transmitting traction, the contact with the hydraulic operating mechanism through the hinge (47), and the follower/slot contact.

A roller control system, according to the present invention as defined in claim 1, arises from further consideration of the full range of constraints which operate upon the tilting of the rollers and the axes about which they tilt, and the resulting conclusion that the orientation of the rollers, and thus the transmitted ratio, can be achieved with a different and simpler design of roller carriages, and constraints upon them and particularly upon the number of contacts between each carriage and adjacent structure. The invention applies particularly to roller control systems in which there is a component of castor angle in the contacts between rollers and races. The invention is defined by the claims, the contents of which are to be read as part of the disclosure of this specification, and the invention will now be described, by way of example, with reference to the following further diagrammatic or schematic drawings in which:-

Figure 4 is a view of one variator taken at right angles to its axis, with some parts shown in section;

Figure 5 shows the same variator, with some parts shown in a section taken on the line V-V in Figure 4;

Figure 6 is a schematic view of parts of another variator;

Figure 7 shows more details of yet another variator, generally in a section taken at right angles to its axis;

Figures 8 to 17 are schematic illustrations of ten different roller control systems according to the invention;

Figure 18 shows the roller, carriage and operating mechanism of another variator, partly in elevation and partly in section;

Figure 19 shows the piston of another operating mechanism;

Figure 20 shows a hydraulic circuit for use in connection with the operating mechanism of Figure 19, and

Figure 21 shows part of an alternative hydraulic circuit.

Figures 4 and 5 show a roller 60 transmitting drive from the input disc 61 to the output disc 62 of a toroidal-race variator housed within a casing 63. Items 61, 60, 62 and 63 correspond essentially with items 4, 20, 13 and 19 of Figure 1. Roller 60 is mounted on an axle 59 in bearings 64, 65 to spin about an axis 58 within a carriage 67 so that both the axis 58 and the roller centre 66 are fixed relative to the carriage, which is itself secured by a threaded connection 68 and locknut 69 to one end of a shaft 70. A double-acting piston 71, mounted on the other end of that shaft, moves within a cylinder 72, and the two chambers 73 and 74 of that cylinder are connected by lines 75 and 76 to pressure fluid source 11 by way of control valve means 43, as in Figure 2. Shaft 70 enters cylinder 72 by way of a flexible sealing gland 77 which is mounted in the cylinder end plate 78 and is capable of limited transverse movement without sealing loss. As is shown best by the magnified insert to Figure 4, a central sealing ring 80 of piston 71 has an outer rim 81 which conforms effectively to the surface of a sphere having as its centre the point 82 which is also the centre of the piston. Piston centre 82 is thus constrained to movement along the axis 84 of cylinder 72, but because of the contouring of rim 81 and the flexible sealing gland 77 the carriage 67 is capable at all times of rotation about axis 84, and of rotation about the orthogonal axes 92 and 93.

Roller 60 contacts the toroidal race 85 of disc 61

at 86, and the corresponding race 87 of disc 62 at 88, and as shown in Figure 4 discs 61 and 62 are rotating as indicated by arrows 89 and 90. According to the invention, we have discovered that the reaction forces active through three points of contact only, namely the two reaction forces between the discs and the roller, and the third reaction force on the piston 71, are between them sufficient to ensure that the roller seeks and holds the appropriate ratio angle at which the piston/fluid and roller/disc reactions are balanced, without any further physical constraint upon roller or carriage. This is of course to be contrasted particularly with the mechanism of US-A-3933054 where, as already explained, four contacts with adjacent mechanism are needed for stability. It should also be noted that there is no requirement for the discs 61, 62 to be capable of simultaneous and equal movements along the CVT axis 2,as is necessary in GB-A-1600972.. The nominal axial position of disc 61 within the CVT will in practice be predetermined, just like that of disc 5 of Figure 1. The essential geometry of a toroidal-race variator requires that roller centre 66 must always lie on the centre circle of the common torus of discs 61 and 62, which in turn lies in the torus mid-plane 91. Therefore the solid angle through which line 83 (which is drawn through roller centre 66 and piston centre 82) can move must be great enough to permit roller centre and torus centre circle to coincide, whatever the position of piston 71 within cylinder 72. Furthermore, because the centre 82 of piston 71 is constrained to follow the fixed axis 84 of cylinder 72, the angle between that axis and the plane 91 therefore becomes the nominal castor angle for the variator. It will however be apparent that the actual castor angle C_1 (Figure 4) lies between plane 91 and line 83, and that this angle will vary slightly in use, dependent upon the position of piston 71 within cylinder 72 and thus of where piston centre 82 lies on axis 84. It should also be noted that in the embodiment of the invention shown in Figure 4, determination of the appropriate ratio angle, as just described, requires that the roller 60 is constrained to spin about a fixed-axis and a fixed centre 66 within carriage 67. If that centre were free to move up and down the spin axis 58, as is the case in some known carriages in this art, a further degree of freedom would exist and the necessary roller control would not be achieved. More particularly, if the spin axis 58 were able to pivot relative to the direction of the control force exerted by piston 71, in the manner permitted by the pin connection 47 in US-A-3933054, a further degree of freedom would again exist and the necessary roller control would not be achieved.

Sufficient ratio control in the embodiment of Figure 4 is thereby achieved by connecting the roller and carriage to operating mechanism (that is to say, the double-acting piston and cylinder 71/72) where only a single point of connection (effectively the piston centre 82) is subject to axial and radial constraint, the carriage being free to move within a limited solid angle having its vertex at that point, and where the entire operating mechanism lies to one side of the roller centre 66. This offers obvious economies in components, compared with the known double-ended carriage restraint provided by ball ends 37, 38 in Figure 2 of the present application, or trunnions 32/recesses 31 in US-A-1865102, and the movable rotors 10, 12, of GB-A-1600972. Another advantage of the embodiment of the present invention shown in Figures 4 and 5 over prior proposals such as that shown in Figures 2 and 3 of the present application is that the single cylinder 72 may conveniently, as shown best in Figure 4, be mounted not on a triangular frame such as item 30 of Figure 2, which like the rollers and carriages must be accommodated between the input and output discs, but directly and simply on the casing 63 of the variator as a whole. This in turn helps to make possible large values of the nominal castor angle between cylinder axis 84 and plane 91, and thus of the actual castor angle C_1 between plane 91 and line 83. Researches indicate that working values of castor angle C_1 of the order of 20° or even more, which are large compared with the castor angles of say 5 - 8° which have been the most frequently used in this art, may promote greater stability in general, and in particular a more prompt return to equilibrium (in which axis 58 and the variator axis 2 intersect) whenever an axial movement of piston 71 has disturbed that equilibrium so as to cause the roller and carriage to tilt about line 83 and so change the transmitted ratio.

In Figures 4 and 5 the entire operating mechanism lies to one side only of the plane which includes the CVT axis 2 and the roller centre 66. This promotes compactness of design, but double-acting piston-and-cylinder combinations like items 71, 72 in Figures 4 and 5 may present problems of construction and operation in certain cases, and Figures 6 and 7 show alternative designs according to the invention, and use the reference numerals of Figures 4 and 5 for all comparable items. In Figure 6 ball members 95 and 96 are mounted on carriage 67 at opposite ends of a diameter, and engage with socket joints in pistons 97, 98 which slide in cylinders 99, 100 respectively. The chambers 101, 102 of these two cylinders are connected to pressure source 11 by way of valve means 43, just like the two chambers of cylinder 72 were in Figure 4. Piston 97 slides accurately within cylinder 99 so that the centre 103 of ball 95 is constrained to the cylinder axis 84, but the flexible sealing ring 105 which spans the annular clearance between piston 98 and its cylinder 100 allows some freedom of movement, with the effect that the guidance of piston 98 by cylinder 100 is as if there was no solid-to-solid contact between the two parts. The only substantial effect upon carriage 67 of the forces exerted upon it by piston 98 are comparable to the pulling forces that piston 97, were it double-acting, would exert upon ball 95. In

this embodiment of the invention ball centre 103 constitutes the effective point of connection between the carriage and its operating mechanism (like piston centre 82 in Figure 4) and the line 106 joining ball centre 103 and roller centre 66 takes the place of line 83 as the line which defines the working castor angle C_1 with the torus mid plane 91. The joint between ball 95 and piston 97 therefore has the characteristic of constraining carriage 67 so that ball centre 103 can move along axis 104, and so that the carriage can also rotate not only about that axis, but also about axis 107 at right angles to it and about the further orthogonal axis that lies perpendicular to both 104 and 107, and thus also to the paper. Cylinders 99 and 100 could both, as shown, be mounted on the variator casing 63.

Figure 7 shows a roller 60, mounted as before in bearings 64, 65 to spin about a fixed centre 66 within a carriage 110, but in this embodiment the carriage 110 is formed with pistons 111 and 112 at opposite ends. These pistons slide within oppositely-facing cylinders 113 and 114 mounted in a frame 115 (comparable to item 30 of Figure 2) located between an input disc 116 and output disc (not shown) of the variator. References 117 and 118 represent the input shaft and variator casing respectively, frame member 115 being fixed to the latter. Ring 119 of piston 111 is shaped like item 80 of piston 71 in Figure 4, so that the centre 120 of the piston has the same function has centre 82 of piston 71, and is constrained to follow the cylinder axis 84. The substantial annular clearance between piston 112 and its cylinder 114 is spanned by a flexible ring 121 comparable to ring 105 of Figure 6, and also as in that Figure the cylinder chambers 101, 102 are connected by way of control valve means 43 to pressure source 11. The effect of piston 112 upon carriage 110 is thus comparable to that of piston 98 in Figure 6, contributing negligible radial constraint and effectively only exerting such force upon the carriage as piston 111, were it double-acting, would do when it pulled. Piston centre 120 therefore constitutes the effective point of connection between the carriage and its operating mechanism, like items 82 and 103 of Figures 4 and 6, and is constrained to move along axis 84 but also permits the carriage 110 to rotate not only about axis 84, but also about orthogonal axis 123 and about a third axis which passes through centre 120 and lies at right angles to both 84 and 123 and therefore also to the paper. In this embodiment of the invention, as in Figure 4, the "nominal" castor angle of the variator will be set by the axis 84 of cylinder 113, but the actual and slightly variable castor angle will in use be that angle, as in Figure 4, at which the line joining roller centre 66 and piston centre 120 intersects the mid-torus plane (not shown, but comparable to item 91 of Figures 4 and 6). As with carriage 67 of Figure 4, carriage 110 is able to move through a limited solid angle about a point constrained to move along axis 84, the angle being sufficient not only to allow the roller 60 to progress through the full range of ratio angles required of the variator, but also of course to permit the roller centre 66 to lie on the torus centre circle (its only geometrically possible position) at all times, whatever the position of piston 111 within its cylinder 113.

While the invention is defined formally by the claims, less formally stated the invention seeks to provide a roller control system which is considerably simplified compared with many known systems, in which the operating mechanism imposes a translational movement upon the roller carriage which determines the location of the roller centre on a circular locus running centrally around the torus, and in which the carriage has sufficient degrees of rotary freedom that nothing constrains the roller centre from following that locus so as to take up the position demanded of it by the operating mechanism at all times. Where the torus is of circular cross-section, as shown in all the Figures, that locus will be the torus centre circle. Figures 8 to 17 illustrate, in outline, only a selection of the types of roller control system that fall within the scope of the present invention. In Figure 8 the piston 131 can both move axially and rotate like a ball within cylinder 130, and is attached rigidly to shaft 132 which is in turn attached rigidly to roller carriage 133. Shaft 132 and carriage 133 can equally well be regarded as together constituting a single, one-piece carriage assembly. Both the centre 134 and the axis of the rotation of roller 135 are fixed relative to the carriage, and the front wall 136 of cylinder 130 can flex to accommodate deflection of rod 132. Cylinder 130 is double-acting, and will in practice be fixed within the transmission. It will be appreciated that the embodiment of the invention, already described in some detail with reference to Figures 4 and 5, is of this type.

In Figure 9, as in Figure 8, piston 131 can rotate as a ball within cylinder 130 and is rigidly attached to rod 132, but now cylinder 130 is only single-acting, and an extension 137 of the rod on the distal side of carriage 133 is connected by a ball joint 138 to a further ball-type piston 139 moving in a single-acting cylinder 140. Piston 139 and cylinder 140 provide the system with the reverse movement that cylinder 130 would provide, were it double-acting as in Figure 8. Figure 9 thus shows a variant of the single-acting systems already described in more detail with reference to Figures 6 and 7.

In Figure 10 cylinder 130 is double-acting again and a piston 141, capable only of translation along and rotation about the cylinder axis, is attached rigidly to a rod 142. The other end of this rod is attached, by way of a ball joint 143, to the carriage 133.

Figure 11 is similar to Figure 10, except that cylinder 130 is now only single-acting. The system is therefore extended by items 137-140, as in Figure 9, to provide the facility for reverse motion.

Figure 12 shows a variation on both Figure 8 and

Figure 10. Cylinder 130 is double-acting as in both of those Figures, piston 141 is as in Figure 10 and its rigid connection 132 with carriage 133 is as in Figure 8. To provide the extra degrees of rotary freedom necessary to the invention, therefore, cylinder 130 is itself mounted to rotate as a ball within a fixed and complementary ball-shaped housing 145. Figure 13, which includes a further piston 146 comparable to item 141 and movable in a cylinder 140 which can rotate like a ball within a housing 147 comparable to 145, represents the corresponding variation of the design of Figure 9.

The system shown in Figure 14 is comparable to that of Figure 10, but the operating mechanism, instead of being in the form of a conventional piston movable within a cylinder, is now an arm 150 rotatable about a fixed centre 151 and attached by a ball joint 152 to rod 132 to which carriage 133 is rigidly fixed as before. Figure 15 is similar, the only difference being the relative positions of roller centre 134 and arm rotation centre 151, and the different shape that the arm 150 must in consequence take.

The system shown in Figure 16 represents yet another variation of that of Figure 8. Cylinder 130 is again double-acting, and carriage 133 is rigidly attached to piston rod 132 as before. However the piston 155 is now flexible, so that whereas the centre 129 (Figure 8) of piston 131 was constrained to follow the axis of cylinder 130, the centre 156 of piston 155 is not. The necessary further constraint upon the system is caused by rod 132 sliding through a matching and sealing cavity in a ball 157 which can rotate without loss of fluid within a matching ball-shaped housing 158 formed in the front wall 159 of cylinder 130.

The system shown in Figure 17 is a variant of that shown in Figure 12. However, whereas in Figure 12 the necessary degrees of rotary freedom were provided by allowing cylinder 130 to rotate bodily, as a ball, within housing 145, in the variant of Figure 17 the necessary two rotary movements are separated. A lug 160 is fixed to cylinder 130, and there is a rotary joint between this lug and a second lug 161 attached to a sleeve 162, this rotary joint allowing mutual rotation about an axis 163. Sleeve 162 can itself rotate about a further, and fixed, axis 164. Axes 163 and 164 are mutually at right angles, but do not intersect.

The embodiments shown in outline in Figures 8 to 17 of the drawings share certain common features. Firstly there are means capable of effecting the translational movement of the roller centre back and forth along the torus centre circle: this is provided by the rotary movement of the ends of the arms 150 in Figures 14 and 15, and by the strokes of the pistons 131, 141 and 155 in the rest of Figures 8 to 17. Secondly, there is freedom for the roller to rotate about a diameter and so to change the transmitted ratio. In Figures 14 and 15 this rotary freedom is provided entirely by the ball joints 152, in Figures 10 and 11 it is provided jointly by the ball joint 143 and by the ability of piston 141 to rotate about the axis of cylinder 130, and in the rest of this group of Figures it is provided by the ability of all of pistons 131, 141 and 155 to rotate about the axes of their respective cylinders 130. Thirdly, both the axis and the centre 134 of rotation of roller 135 are fixed relative to the rigid carriage 133.

Fourthly, the position of the roller centre, which is constrained to follow the torus centre circle, imposes no loads on the carriage nor on its operating mechanism. This position may be affected by dimensional variations in manufacture or assembly : in the plane of the torus centre circle the roller centre will follow the curve of the torus centre circle, while the torus centre circle itself will move in the direction of the transmission axis under the influence of end-load changes. Avoidance of such loads is achieved by designing the carriage and its operating mechanism to give the roller centre freedom to move through two intersecting arcs which lie in different and intersecting planes. This mechanism will also accommodate the small changes in castor angle which occur in use in the embodiments of Figs. 8, 9, 12, 13, 16 and 17.

Fifth, in all the embodiments of the invention there is a constant and triangular relationship between the two points at which the roller reacts tangentially with the discs, and the location at which the control force is applied to the roller assembly. In all of Figures 8 to 17, the control force acts along rod 132, and is applied to that rod at a location displaced from the roller axis. The two roller/disc contacts are fixed relative to that rod, but are displaced from it and from each other. A stable triangle of forces, acting in the same plane, is therefore established. This stable triangle would still exist even if the centre and axis of the roller were displaced from, but still fixed relative to, the axis of rod 132, because the geometry of the resolution of the control force and of the two tangential roller/disc reaction forces relative to the triangle would still be constant.

In the embodiment of Figure 18, a roller 60 transmits traction between the part-toroidal race 85 of an input disc 61 and the corresponding part-toroidal race 87 of an output disc 62, and is mounted in a carriage 67, fixed rigidly at one end (as in Figure 4) to a piston 71, the centre 82 of which is constrained to follow the axis 84 of the cylinder 190 in which it moves. The opposite end of carriage 67 is formed as a spherical face 191 which in use abuts the flat working face 192 of a separate piston 193 moveable within a cylinder 194. Carriage 67 and piston 193 are thus not only separate, but also without any interlock in use, and are therefore unlike the designs shown in Figure 6, where there is a ball-in-socket joint between carriage 67 and piston 98 and in Figures 9, 11 and 13 where carriage 133 is also fixed to pistons 139 or 146. The construction of Figure 18 has the advantage that roller 60 can be put in place between discs 61 and 62 by the follow-

ing succession of simple steps :-

1. With fluid power disconnected, depress piston 193 within cylinder 194;

2. Insert piston 71 into cylinder 190, and introduce roller 60 between discs 61 and 62 until the crown of face 191 is substantially aligned with the axis of piston 193;

3. Release piston 193. Then, when the CVT starts to operate, and input disc 61 rotates and pistons 71 and 193 are exposed to fluid pressure, roller 60 will take up the correct orientation and working face 192 will abut spherical face 191 to exert the restoring force which single-acting piston 71 itself cannot exert.

Furthermore, the inner ends of cylinders 190, 194 are not closed, but the full bores of these cylinders are in communication with annular galleries 200, 201 formed within the casing 63 of the CVT. Galleries 200, 201 are each in communication with fluid source 11 and associated with control valve means 43, as indicated schematically in Figures 4 and 6, and also communicate with return lines 202 and 203 as customary. If carriage 67 makes a sudden axial movement in use due to some emergency such as a crash stop or a change in road surface from normal to icy for example, the large cross-section of the access between the galleries 200, 201 and the cylinders facilitates a speedy entry of fluid into cylinder 190 and exit of fluid from cylinder 194, or vice versa. Such a rapid entry or exit fluid could be impossible, and give rise to undesirable back pressures, if the cylinders were blind-ended, and connected to the pressure fluid circuit by way only of the normal small-bore inlet and outlet ports.

Where a set of rollers (such as items 20, Figure 1) transmit torque between the same input and output disc, the same galleries 200, 201 may conveniently communicate with the corresponding pistons 71, 193 of all rollers in the set. Where there are two sets of rollers (like items 20, 25 in the same Figure) used together in a double-ended CVT, galleries 200, 201 may communicate with the pistons of one set and galleries 200a and 201a also formed within casing 63 may communicate with the pistons of the other set. Galleries 200, 200a are connected by a conduit 217 so that the pressures existing within them are equal, and galleries 201, 201a are connected to like effect by a conduit 218.

Piston 71, moving within cylinder 190 in Figure 19 of the drawings, is modified by an extension 205, the end face 206 of which meets the side wall 207 at a smoothly curved edge 208. The edge of return line 202, where it enters gallery 200, is formed with an angled face at 210. If in use of the CVT an overload/abuse condition develops and is reflected in an extreme axial movement (leftwards, as in Figure 19) of piston 71, surfaces 208 and 210 will approach, so imposing an extra resistance on the normal flow of fluid out of gallery 200 into return line 202. Pressure in galleries 200 and 200a will therefore rise, and since the pressure in those galleries operates on all the other pistons 71, all those other pistons will be subjected to an enhanced force which opposes them as they approach the corresponding extremes of their ranges of axial movement. A "hydraulic end stop" effect is therefore set up. Figure 20 diagrammatically shows one possible practical embodiment in which the fluid source 11 comprises twin pumps 215, 216 connected to the galleries 200, 200a, 201, 201a feeding the operating cylinders 190, 194 of the two sets of cylinders (20, 25) of a double-ended CVT of the toroidal-race, rolling-traction type. The return lines 202, 203 of the hydraulic circuit connect with the cylinders of only one roller (60a), which acts as the "master" for all the other rollers. Piston 71 of the carriage 67 of this roller carries an extension 205, which exercises an "end stop" action as already described when piston 71 tends to overshoot within cylinder 190, and piston 193 of that same roller carries a seal 220 which exercises a similar end stop effect, by approaching cylinder end wall 221 and obstructing outlet port 222, when that piston tends to overshoot within cylinder 194. It should also be noted that the end load cylinders 223, which generates the end load force urging discs 61, 62 into contact with rollers 60, is connected to a part of the hydraulic circuit which is close to the galleries and to the roller operating mechanisms, with no substantial resistance intervening, so that equal pressures exist in the galleries and cylinder 223 at all times.

Generating the end stop effect by means of the principal piston (71) of the roller operating mechanism may require accurate machining of the edge 208, which must conform to part of the surface of a sphere with centre 82. Figure 21 shows part of an alternative design in which the simpler "second piston" 193 of one roller (60b) generates the end stop effect in the left-hand side of the hydraulic circuit when that piston tends to overshoot. The adjacent roller 60c is arranged the other way round so that its "second" piston 193 is on the right-hand side of the circuit (corresponding to galleries 201, 201a), and generates the end stop effect on that side of the circuit when it overshoots.

Claims

1. A roller control system for a continuously variable transmission (CVT) of the toroidal race, rolling traction type comprising a roller assembly which includes a carriage (67) and bearings (64, 65) mounted thereon and a roller (60) mounted to spin in those bearings, in which the roller contacts and transmits torque between coaxial rotatable discs (61, 62) presenting input and output races (85, 87) conforming to different parts of the surface of a single torus and in so doing is subjected

to traction forces at the disc-roller contacts, and comprising an operating mechanism which includes fixed parts (72) and movable parts (71), the movable parts being arranged with a castor angle relative to the discs and reciprocable over a predetermined stroke of operating movement and operable to apply a predetermined control force to the carriage, in which the operating mechanism and the carriage seek an equilibrium position in which the resultant of the control force and of the traction forces experienced by the roller assembly in a plane at right angles to the axis of the discs is zero, in which the carriage includes a rigid structure relative to which the roller axis (58) is fixed and which contacts the operating mechanism at a location (82) displaced from the roller axis so that the roller assembly is located by only three contacts with adjacent components, namely the two disc/roller contacts (86, 88) and the contact (82) with the operating mechanism **characterised**

in that the position of the roller centre (66) along the roller axis is fixed,

and in that the roller assembly is rotatable about more than one axis relative to the fixed parts of the operating mechanism.

2. A roller control system according to Claim 1 in which the contact between the operating mechanism and the roller assembly is made by way of a ball joint (95, Fig. 6) or the like permitting mutual rotation about more than one axis.

3. A roller control system according to Claim 1 in which the operating mechanism a piston moveable within a cylinder.

4. A roller control system according to Claim 3 in which the piston and cylinder combination (71, 72) is double-acting.

5. A roller control system according to Claim 3 in which the piston and cylinder combination (97, 99 Figure 6) is single-acting and capable of exerting a force in a first direction, and a second piston and cylinder (98, 100) are provided capable of exerting a force in a second and substantially opposite direction.

6. A roller control system according to Claim 3 in which the piston (155, Fig. 16) is flexible, and the piston centre (156) has freedom to depart from the cylinder axis.

7. A roller control system according to Claim 3 in which the relative rotation between the piston and the cylinder (141, 130, Fig. 10) takes place about the cylinder axis only, and in which the contact between the roller assembly and the operating mechanism is made by way of a further joint (143) permitting rotation about other axes.

8. A roller control system according to Claim 1 in which the operating mechanism is located to one side only of the plane which includes the CVT axis (2, Figure 4) and the roller centre (66).

9. A roller control system according to Claim 1 in which the operating mechanism includes a single-action piston-and-cylinder combination (71, 190, Fig. 18) comprising a first piston movable within a first cylinder and capable of exerting a force in a first direction, in which the operating mechanism also includes a second piston and cylinder (193, 194) capable of exerting a force in a second and substantially opposite direction, and in which the second piston and the carriage (67) are separate items which abut each other in use in a non-interlocking manner.

10. A roller control system according to Claim 1 including a hydraulic piston-and-cylinder combination (71, 190, Figure 18) by which the control force is applied to the carriage, and a port formed in the cylinder by which the cylinder is in communication with a hydraulic circuit (200), in which the cross-section of the port coincides with substantially the full bore of the cylinder itself.

11. A roller control system according to Claim 1 in which the operating system includes a hydraulic operating circuit and at least one piston-and-cylinder combination in communication with that circuit, and in which the communication (206, 208, 210, Fig. 19) between cylinder and circuit is such that approach of the piston towards an end of its permitted stroke, thus indicating an "overload" or other emergency condition of the system, obstructs the circuit, so causing fluid pressure rise upstream of the obstruction and opposing further piston overshoot.

12. A continuously variable transmission (CVT) of the toroidal race, rolling traction type, including a roller control system according to Claim 1.

13. A CVT according to Claim 12 including a fixed casing structure (63), characterised in that at least part of the operating mechanism is mounted on the fixed casing structure.

14. A continuously variable transmission (CVT) of the toroidal race, rolling type including a roller control system according to Claim 10, including a CVT casing, and in which the hydraulic circuit includes a gallery (200,201) of ring-like shape formed

within the CVT casing, and coaxial with the main CVT axis.

15. A CVT according to Claim 14 in which the gallery conforms to the shape of only an incomplete ring, the break in the ring allowing access for other components to pass through the CVT casing.

Patentansprüche

1. Rollensteuerungssystem für ein stufenloses Getriebe (CVT) vom Schwertrollentyp; mit einer Rollenbaugruppe, die einen Käfig (67) und daran montierte Lager (64, 65) sowie eine in diesen Lagern drehbar gelagerte Rolle (60) aufweist, wobei die Rolle die koaxialen drehbaren Scheiben (61, 62) mit Antriebs- und Abtriebslaufringen (85, 87), die mit verschiedenen Teilen der Oberfläche eines einzigen Torus zusammenfallen, berührt und zwischen ihnen ein Drehmoment überträgt und dabei Antriebskräften an den Kontaktstellen zwischen Scheiben und Rolle ausgesetzt ist; sowie mit einem Betätigungsmechanismus, der feste Teile (72) und bewegliche Teile (71) aufweist, wobei die beweglichen Teile mit einem Nachlaufwinkel in bezug auf die Scheiben angeordnet sind, über eine vorgegebene Hublänge der Betätigungsbewegung hin und her bewegt werden können und so arbeiten, daß sie eine vorgegebene Steuerkraft auf den Käfig ausüben; wobei der .Betätigungsmechanismus und der Käfig eine Gleichgewichtslage anstreben, in der die Resultierende aus der Steuerkraft und den in einer zur Scheibenachse senkrechten Ebene auf die Rollenbaugruppe einwirkenden Antriebskräften gleich Null ist, wobei der Käfig eine starre Struktur aufweist, bezüglich derer die Rollenachse (58) fixiert ist und die den Betätigungsmechanismus an einer gegen die Rollenachse versetzten Stelle (82) berührt, so daß die Rollenbaugruppe durch nur drei Kontaktstellen mit benachbarten Baugruppen fixiert wird, nämlich die beiden Kontaktstellen (86, 88) zwischen Scheiben und Rolle und die Kontaktstelle (82) mit dem Betätigungsmechanismus;

dadurch gekennzeichnet, daß

die Position des Rollenmittelpunkts (66) entlang der Rollenachse fixiert ist; und

die Rollenbaugruppe bezüglich der festen Teile des Betätigungsmechanismus um mehr als eine Achse drehbar ist.

2. Rollensteuerungssystem nach Anspruch 1, wobei der Kontakt zwischen dem Betätigungsmechanismus und der Rollenbaugruppe über ein Kugelgelenk (95, Fig. 6) oder dergleichen erfolgt, das eine gegenseitige Drehung um mehr als eine Achse zuläßt.

3. Rollensteuerungssystem nach Anspruch 1, wobei der Betätigungsmechanismus einen in einem Zylinder beweglichen Kolben aufweist.

4. Rollensteuerungssystem nach Anspruch 3, wobei die Kombination aus Kolben und Zylinder (71, 72) doppeltwirkend ist.

5. Rollensteuerungssystem nach Anspruch 3, wobei die Kombination aus Kolben und Zylinder (97, 99, Fig. 6) einfachwirkend ist und eine Kraft in einer ersten Richtung ausüben kann und eine zweite Kolben-Zylinder-Kombination (98, 100) vorgesehen ist, die eine Kraft in einer zweiten, im wesentlichen entgegengesetzten Richtung ausüben kann.

6. Rollensteuerungssystem nach Anspruch 3, wobei der Kolben (155, Fig. 16) flexibel ist und der Kolbenmittelpunkt (156) von der Zylinderachse abweichen kann.

7. Rollensteuerungssystem nach Anspruch 3, wobei die relative Drehung zwischen Kolben und Zylinder (141, 130, Fig. 10) nur um die Zylinderachse erfolgt und der Kontakt zwischen der Rollenbaugruppe und dem Betätigungsmechanismus durch ein weiteres Gelenk (143) hergestellt wird, das eine Drehung um andere Achsen zuläßt.

8. Rollensteuerungssystem nach Anspruch 1, wobei der Betätigungsmechanismus auf nur einer Seite der Ebene angeordnet ist, welche die CVT-Achse (2, Fig. 4) und den Rollenmittelpunkt (66) enthält.

9. Rollensteuerungssystem nach Anspruch 1, wobei der Betätigungsmechanismus eine einfachwirkende Kolben-Zylinder-Kombination (71, 190, Fig. 18) mit einem ersten, in einem ersten Zylinder beweglichen Kolben, die eine Kraft in einer ersten Richtung ausüben kann, und außerdem eine zweiten Kolben und einen zweiten Zylinder (193, 194) aufweist, die eine Kraft in einer zweiten und im wesentlichen entgegengesetzten Richtung ausüben können, und wobei der zweite Kolben und der Käfig (67) voneinander getrennte Elemente sind, die im Gebrauch in nichtverriegelnder Weise aneinanderstoßen.

10. Rollensteuerungssystem nach Anspruch 1 mit einer Kombination aus Hydraulikkolben und -zylinder (71, 190, Fig. 18), durch welche die Steuerkraft auf den Käfig ausgeübt wird, und einer im Zylinder ausgebildeten Öffnung, durch welche der Zylinder mit einem Hydraulikkreislauf (200) in

Verbindung steht, wobei der querschnitt der Öffnung im wesentlichen mit der vollen Bohrung des Zylinders selbst zusammenfällt.

11. Rollensteuerungssystem nach Anspruch 1, wobei das Betätigungssystem einen hydraulischen Betätigungskreislauf und mindestens eine mit diesem Kreislauf verbundene Kolben-Zylinder-Kombination aufweist und wobei die Verbindung (206, 208, 210, Fig. 19) zwischen Zylinder und Kreislauf so beschaffen ist, daß bei einer Annäherung des Kolbens an ein Ende seiner zulässigen Hublänge, die auf eine "Überlastung" oder einen anderen Notfallzustand des Systems hindeutet, der Kreislauf blockiert wird, was zu einem Anstieg des Flüssigkeitsdrucks in Strömungsrichtung vor der Blockierung führt und einer weiteren Hubwegüberschreitung des Kolbens entgegenwirkt.

12. Stufenlos regelbares Getriebe (CVT) vom Schwertrollentyp, das ein Rollensteuerungssystem nach Anspruch 1 aufweist.

13. CVT nach Anspruch 12, mit einer festen Gehäusekonstruktion (63), dadurch gekennzeichnet, daß zumindest ein Teil des Betätigungsmechanismus an der festen Gehäusekonstruktion montiert ist.

14. Stufenlos regelbares Getriebe (CVT) vom Schwertrollentyp, das ein Rollensteuerungssystem nach Anspruch 10 aufweist, mit einem CVT-Gehäuse, wobei der Hydraulikkreislauf einen ringförmigen Tunnel (200, 201) aufweist, der innerhalb des CVT-Gehäuses ausgebildet ist und koaxial zur Hauptachse des CVT verläuft.

15. CVT nach Anspruch 14, wobei der Tunnel die Form eines nur unvollständigen Rings hat und die Ringunterbrechung den Durchgang anderer Bauelemente durch das CVT-Gehäuse ermöglicht.

Revendications

1. Dispositif de commande de galet pour une transmission continûment variable (CVT) à piste toroïdale du type à traction par roulement comprenant un ensemble de galet comprenant un chariot (67) et des paliers (64, 65) montés dessus et un galet (60) monté pour tourner dans ces paliers, ensemble dans lequel le galet est en contact et transmet le couple entre des disques tournant coaxiaux (61, 62) présentant des pistes d'entrée et de sortie (85, 87) s'adaptant à différentes parties de la surface d'un seul tore et ainsi est soumis à des efforts de traction aux contacts disque/galet et comprenant un mécanisme d'actionnement qui comprend des parties fixes (72) et des parties mobiles (71), les parties mobiles étant disposées avec un angle d'orientation par rapport aux disques et pouvant se déplacer alternativement sur une course prédéterminée du déplacement d'actionnement et actionnables pour appliquer une force de commande prédéterminée au chariot, dans lequel le mécanisme d'actionnement et le chariot recherchent une position d'équilibre dans laquelle la résultante de la force de commande et celles de traction, subies par l'ensemble de galet dans un plan à angle droit par rapport à l'axe des disques, est nulle, dans lequel le chariot comprend une structure rigide par rapport à laquelle l'axe de galet (58) est fixe et en contact avec le mécanisme d'actionnement en un lieu (82) décalé de l'axe de galet de façon à ce que l'ensemble de galet soit localisé par seulement trois contacts avec des composants adjacents, en fait les deux contacts disque/galet (86, 88) et le contact (82) avec le mécanisme d'actionnement,

dispositif **caractérisé** en ce que la position du centre de galet (66) le long de l'axe de galet est fixe et en ce que l'ensemble de galet peut tourner selon plus d'un axe par rapport aux parties fixes du mécanisme d'actionnement.

2. Dispositif de commande de galet selon la revendication 1, dans lequel le contact entre le mécanisme d'actionnement et l'ensemble de galet est réalisé au moyen d'un joint à rotule (95, figure 6) ou similaires permettant une rotation mutuelle autour de plus d'un axe.

3. Dispositif de commande de galet selon la revendication 1, dans lequel le mécanisme d'actionnement comprend un piston mobile dans un cylindre.

4. Dispositif de commande de galet selon la revendication 3, dans lequel la combinaison du piston et du cylindre (71, 72) est à double effet.

5. Dispositif de commande de galet selon la revendication 3, dans lequel la combinaison du piston et du cylindre (97, 99, figure 6) est à simple effet et peut exerce: un effort dans une première direction et une second combinaison de piston et cylindre (98, 100) est prévue, pouvant exercer un effort dans une seconde direction globalement opposée.

6. Dispositif de commande de galet selon la revendication 3, dans lequel le piston (155, figure 16) est flexible et le centre de piston (156) a la liberté de se décaler de l'axe de cylindre.

7. Dispositif de commande de galet selon la revendication 3, dans lequel la rotation relative entre le piston et le cylindre (141, 130, figure 10) prend place seulement selon l'axe du cylindre et dans lequel le contact entre l'ensemble de galet et le mécanisme d'actionnement est réalisé au moyen d'un joint supplémentaire (143) permettant une rotation selon d'autres axes.

8. Dispositif de commande de galet selon la revendication 1, dans lequel le mécanisme d'actionnement n'est situé que sur un côté du plan contenant l'axe de la CVT (2, figure 4) et le centre de galet (66).

9. Dispositif de commande de galet selon la revendication 1, dans lequel le mécanisme d'actionnement comprend une combinaison piston/cylindre à simple effet (71, 190, figure 18) comprenant un premier piston mobile dans un premier cylindre et pouvant exercer un effort dans une première direction, dans lequel le mécanisme d'actionnement comprend aussi un second piston et cylindre (193, 194) pouvant exercer un effort dans une seconde direction globalement opposée et dans lequel le second piston et le chariot (67) sont des pièces séparées en butée l'une contre l'autre en fonctionnement sans blocage mutuel.

10. Dispositif de commande de galet selon la revendication 1, comprenant une combinaison piston/cylindre hydraulique (71, 190, figure 18) par laquelle la force de commande est appliquée au chariot et un orifice formé dans le cylindre par lequel le cylindre est en communication avec un circuit hydraulique (200), dans lequel la section droite de l'orifice coïncide avec le plein alésage du cylindre lui-même.

11. Dispositif de commande de galet selon la revendication 1, dans lequel le dispositif d'actionnement comprend un circuit hydraulique d'actionnement et au moins une combinaison piston/cylindre en communication avec ce circuit et dans lequel la communication (206, 208, 210, figure 19) entre le cylindre et le circuit est telle qu'une approche du piston vers une extrémité de sa course permise, indiquant alors une "surcharge" ou une autre condition d'urgence du dispositif, obstrue le circuit, provoquant ainsi une montée de la pression hydraulique en amont de l'obstruction et s'opposant, de plus, au dépassement du piston.

12. Transmission continûment variable (CVT) à piste toroïdale du type à traction par roulement comprenant un dispositif de commande de galet selon la revendication 1.

13. Transmission continûment variable (CVT) selon la revendication 12, comprenant une structure fixe de carter (63), caractérisée en ce qu'au moins une partie du mécanisme d'actionnement est montée sur la structure fixe du carter.

14. Transmission continûment variable (CVT) à piste toroïdale, du type à traction par roulement comprenant un dispositif de commande de galet selon la revendication 10, comprenant un carter de CVT et dans laquelle le circuit hydraulique comprend une galerie (200, 201) en forme d'anneau formée dans le carter de la CVT et coaxiale à l'axe principal de la CVT.

15. Transmission continûment variable (CVT) selon la revendication 14, dans laquelle la galerie ne prend la forme que d'un anneau incomplet, l'interruption dans l'anneau autorisant l'accès à d'autres composants pour traverser le carter de la CVT.

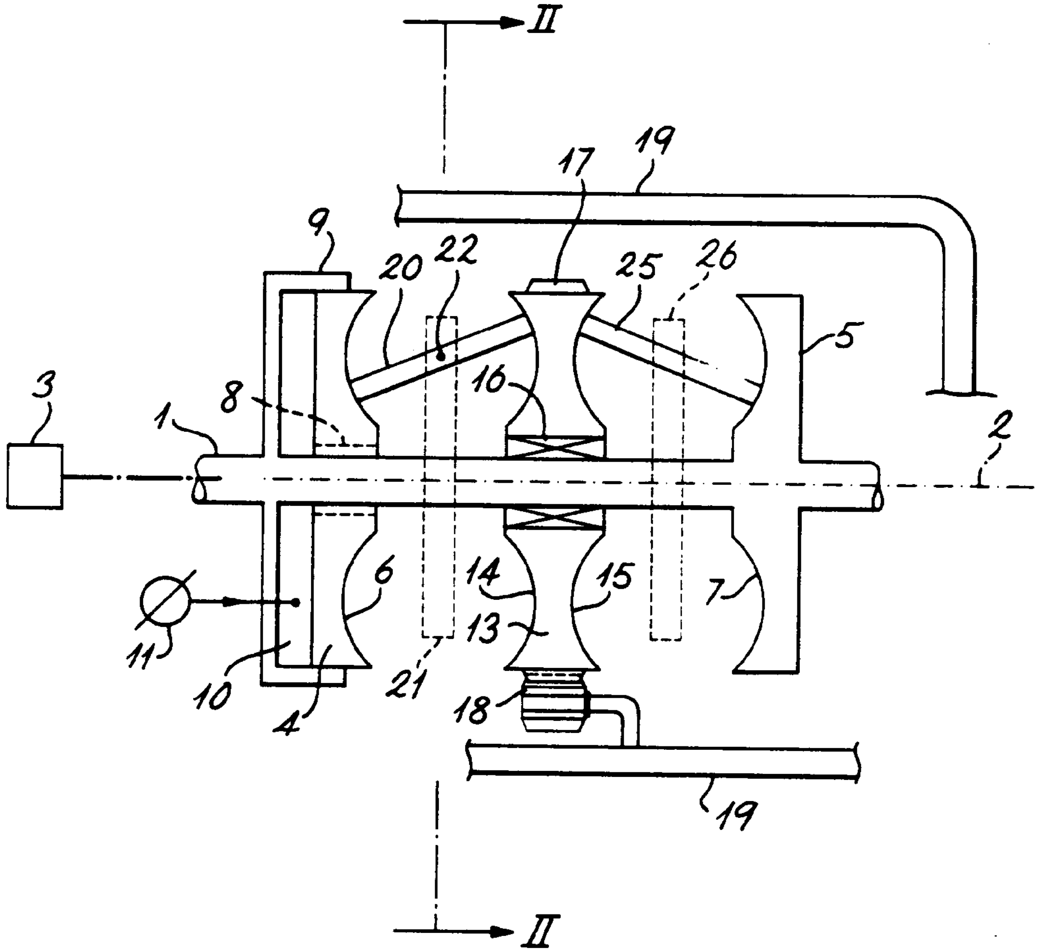

Fig. 1

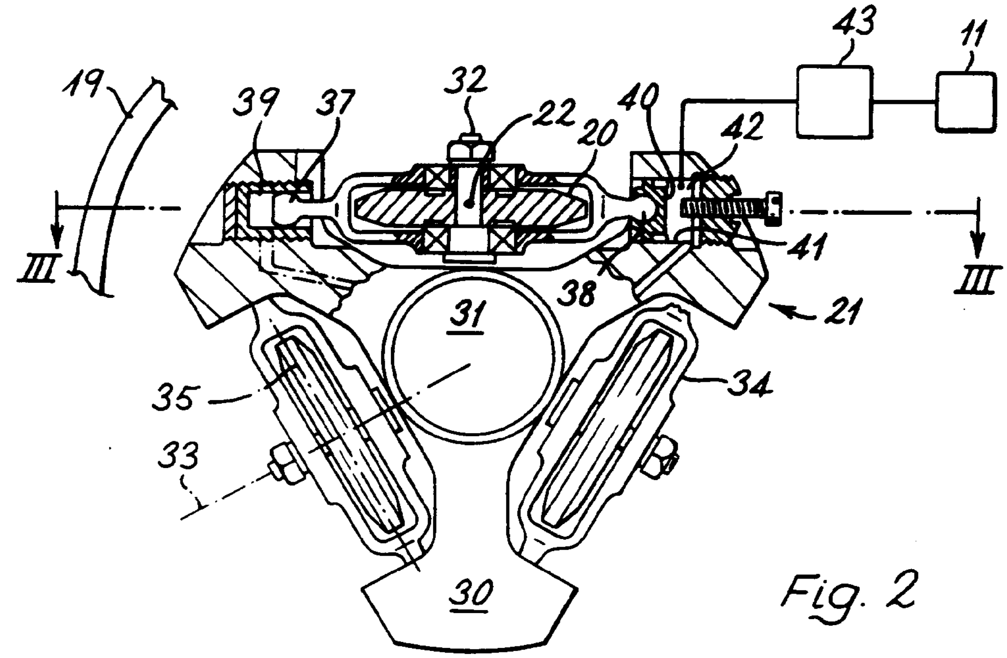
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Fig. 2

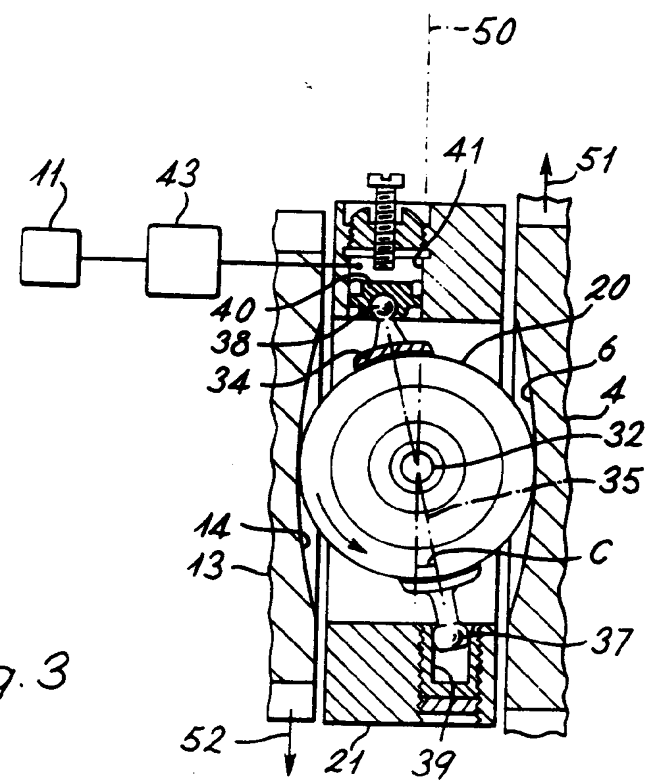
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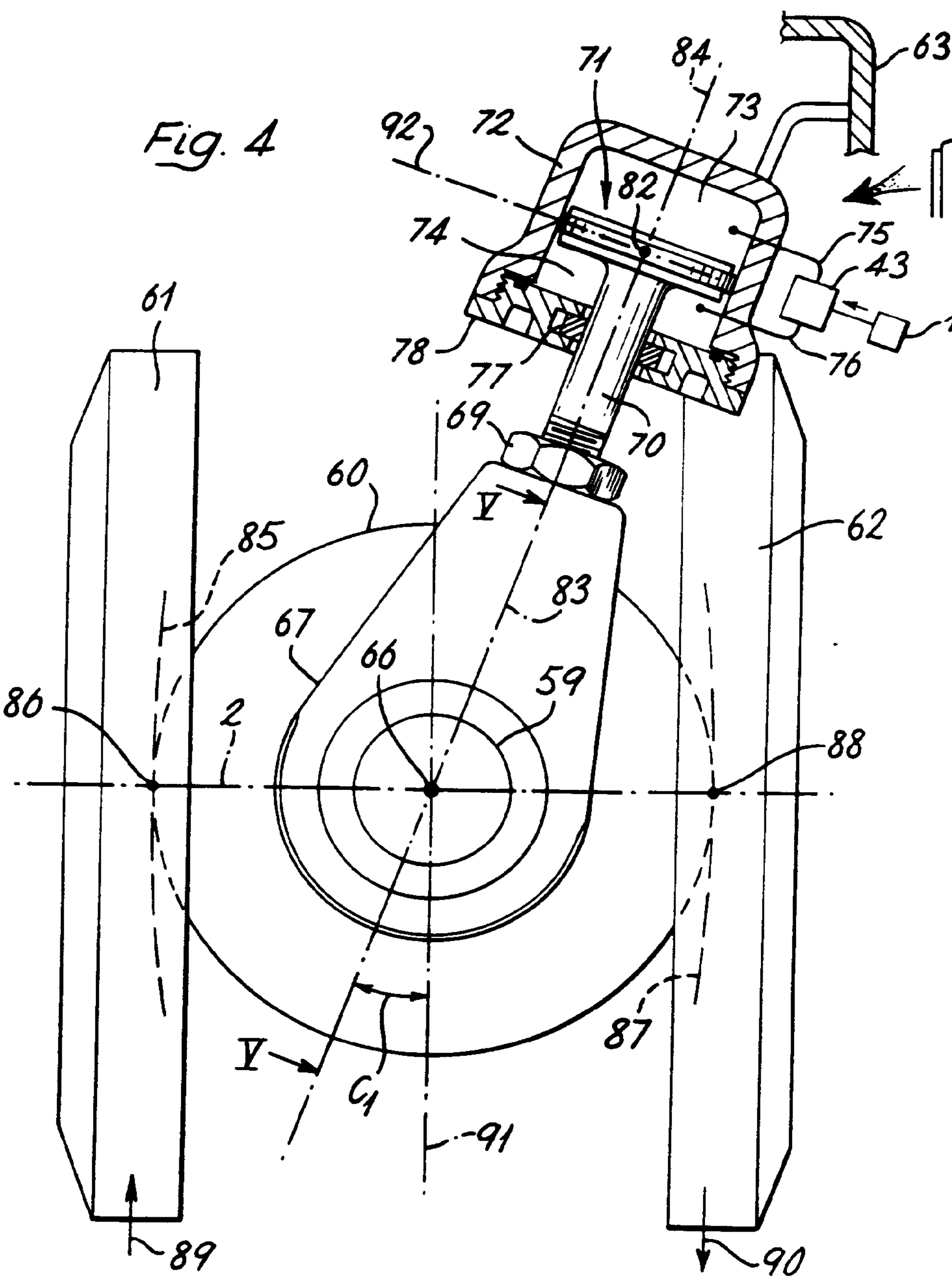
Fig. 3

Fig. 4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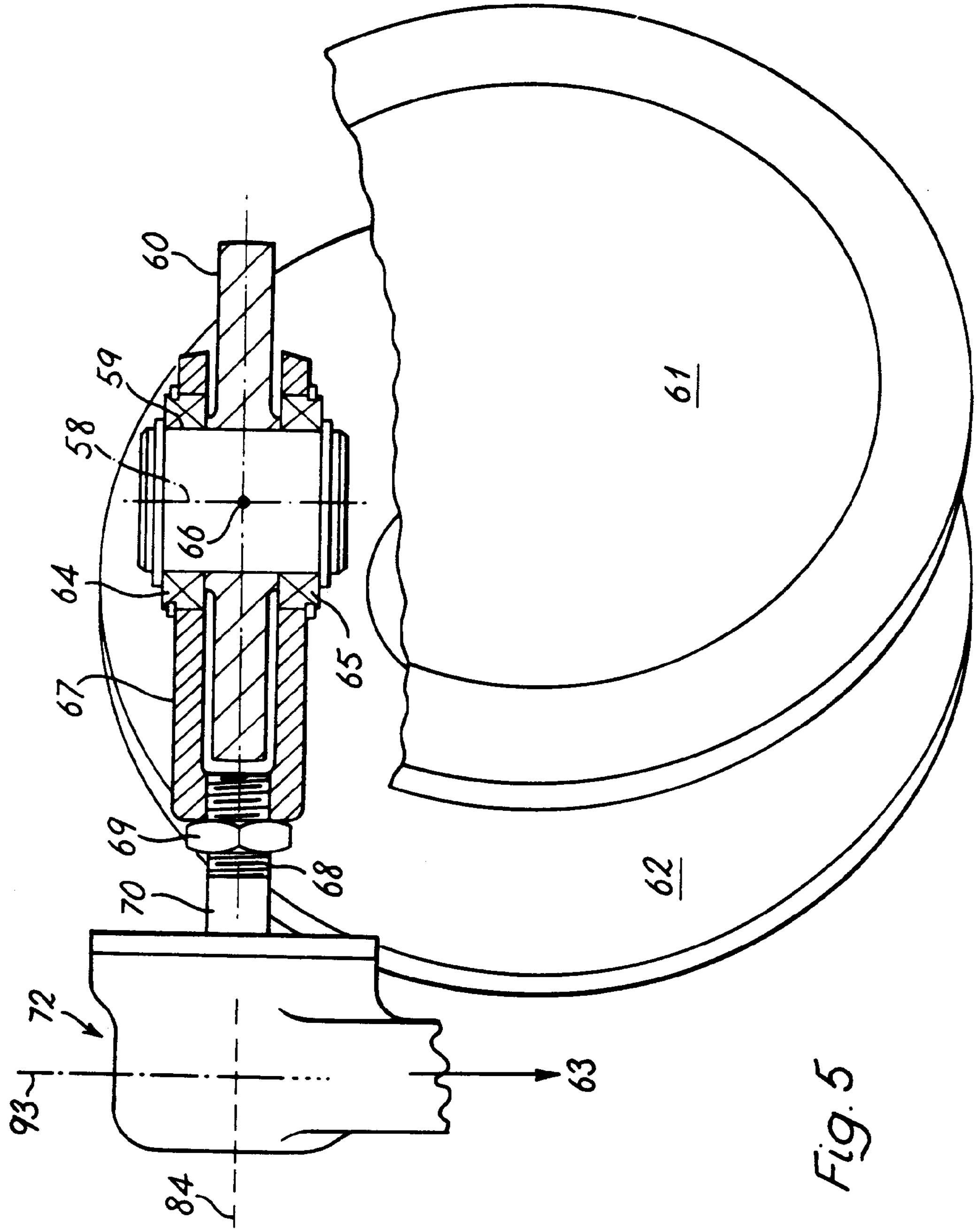

Fig. 5

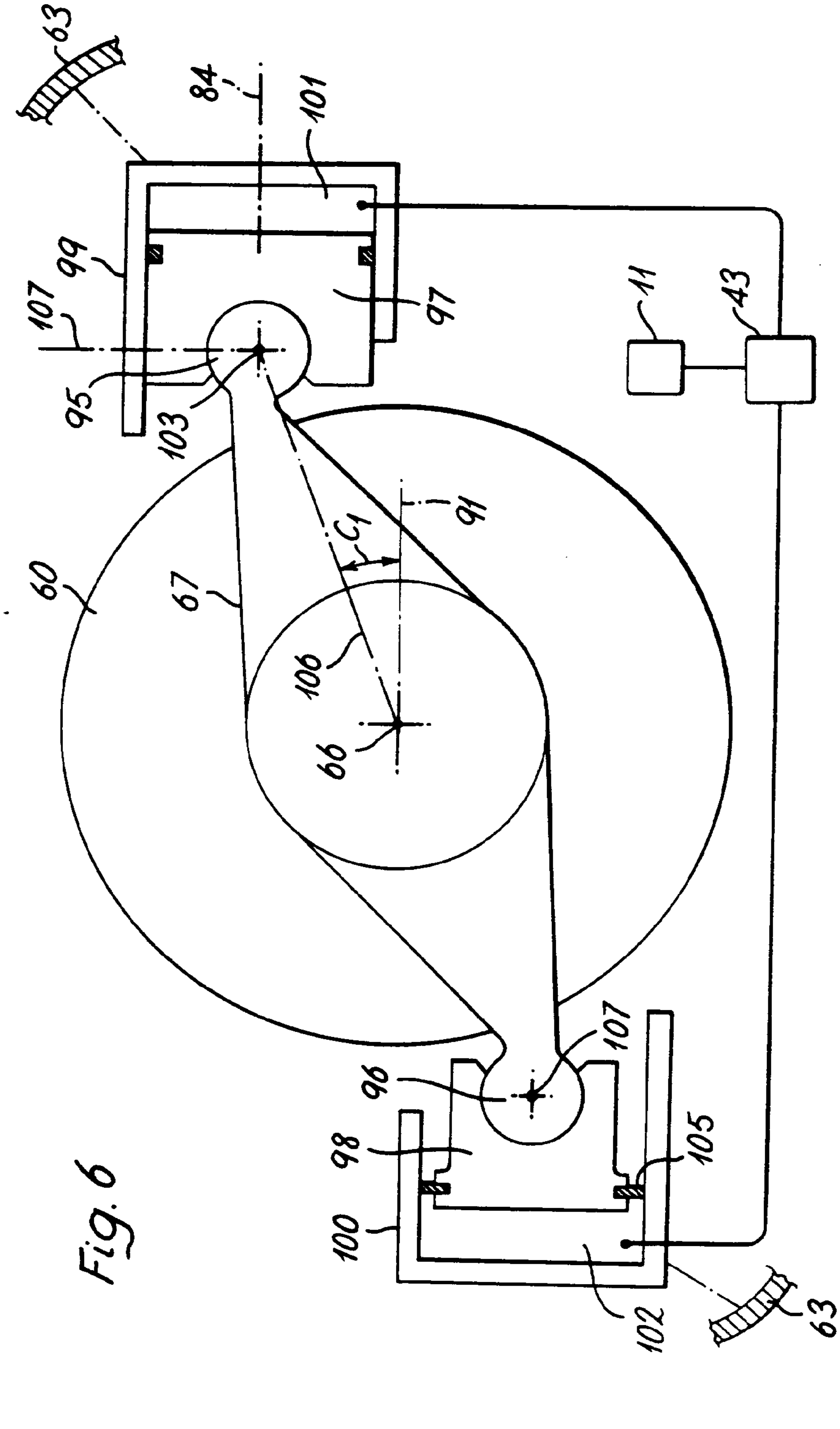
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Fig. 6

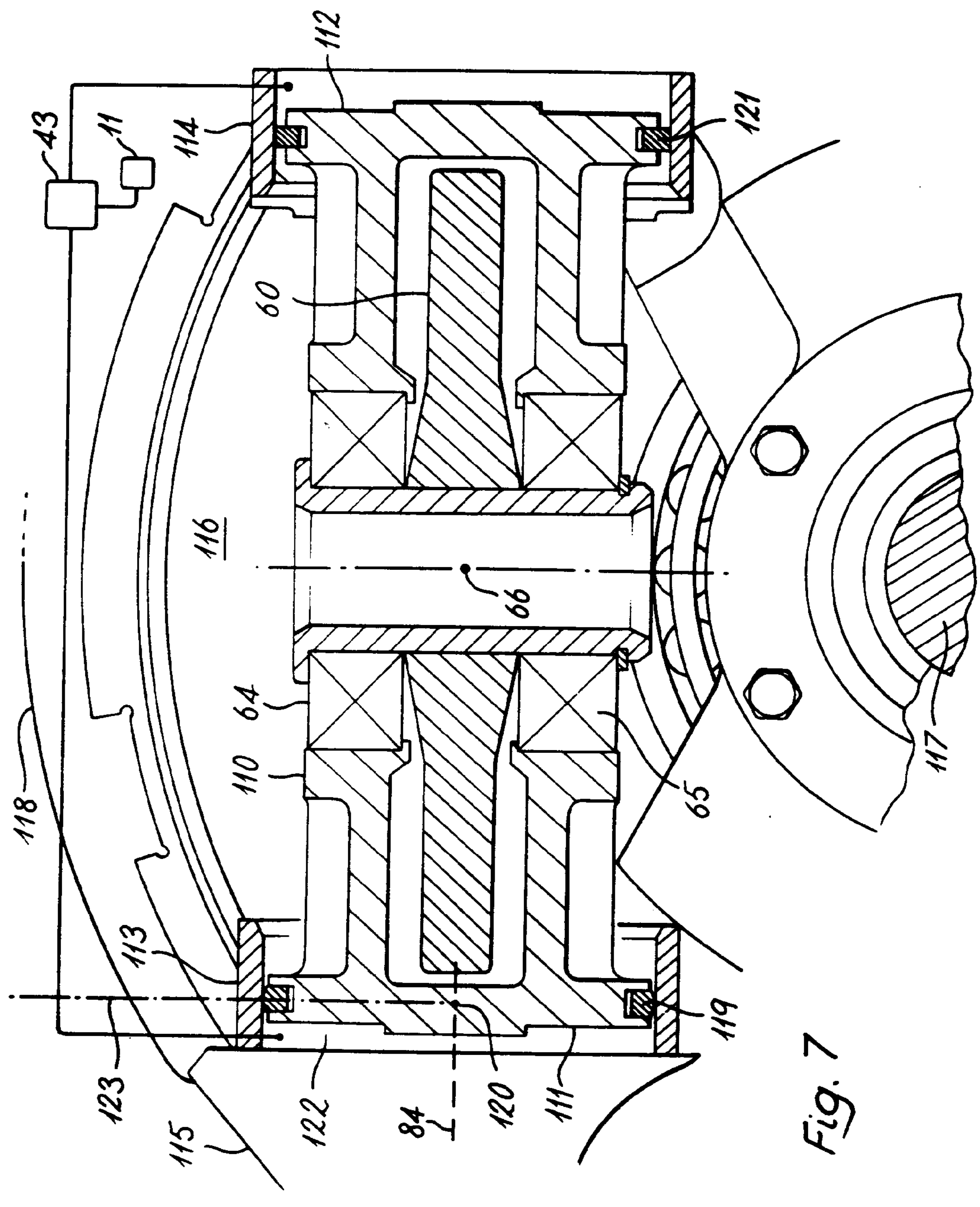
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Fig. 7

Fig. 8

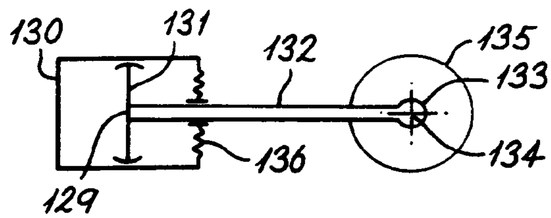


Fig. 9

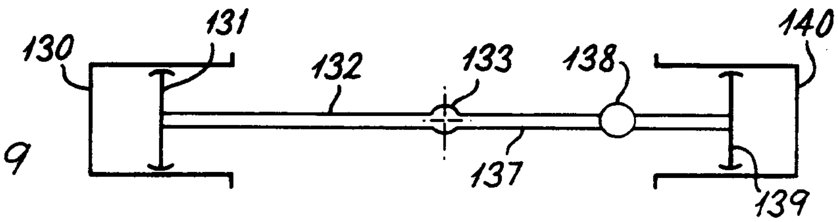


Fig. 10

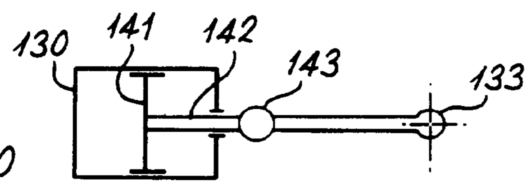


Fig. 11

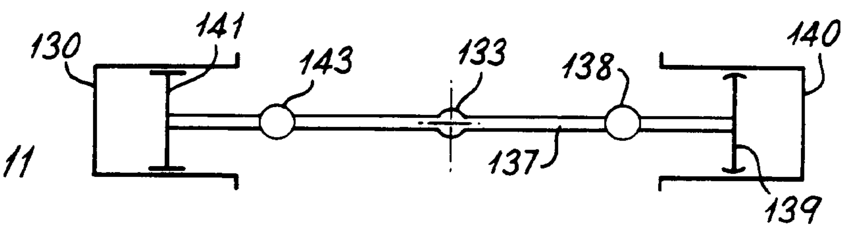


Fig. 12

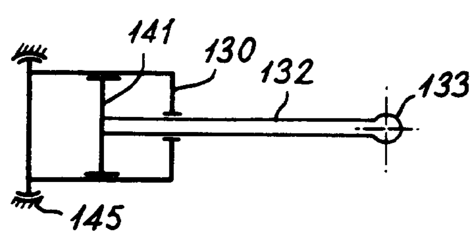
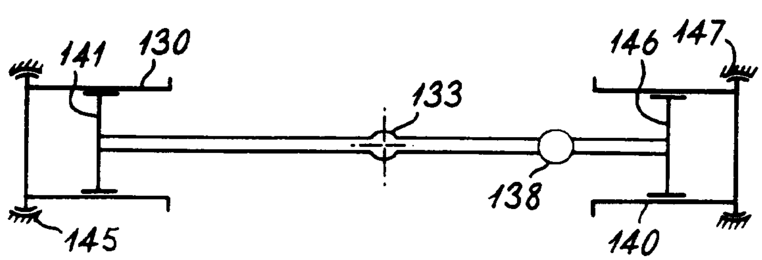


Fig. 13



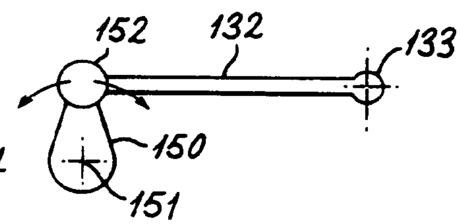


Fig. 14

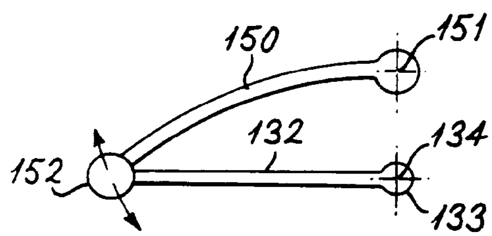


Fig. 15

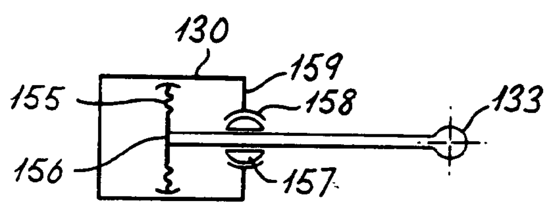


Fig. 16

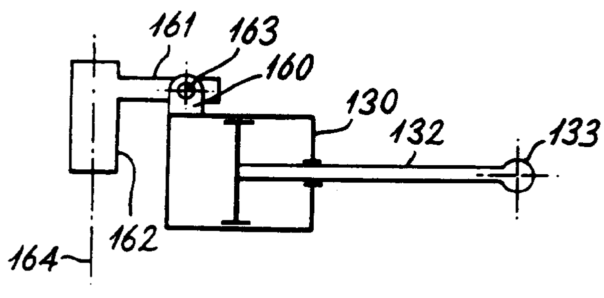
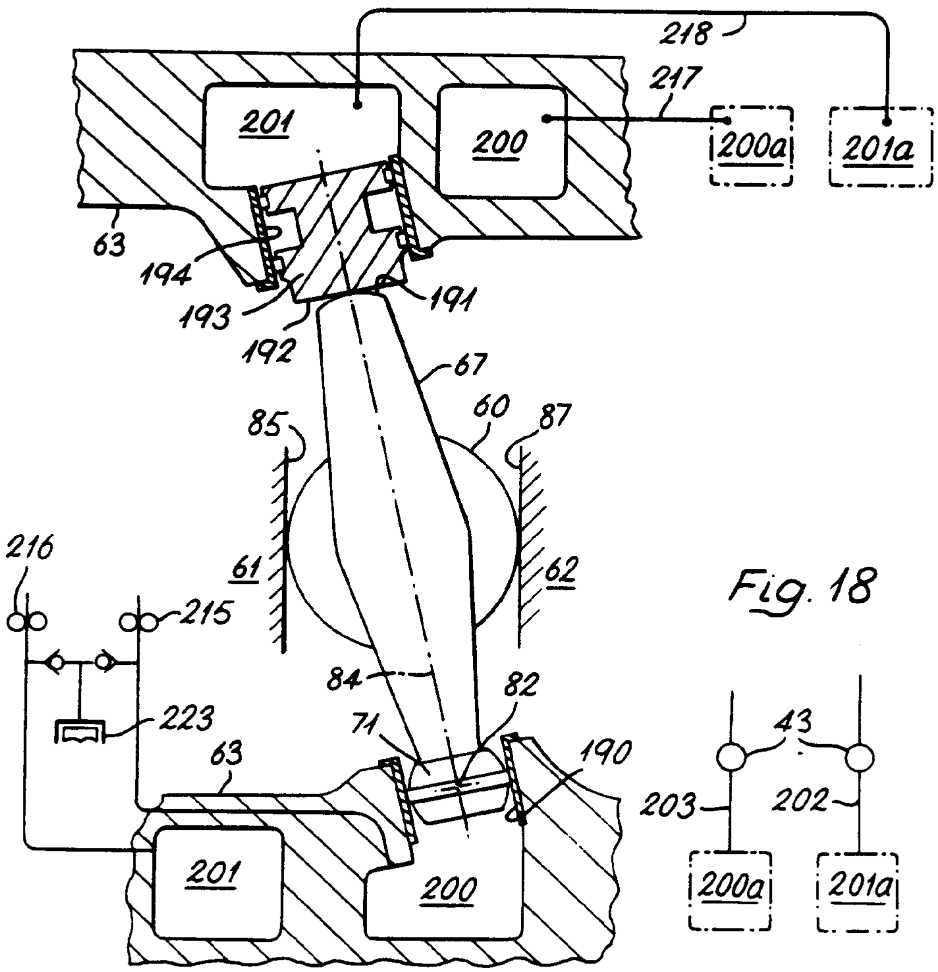


Fig. 17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43
203
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200
200a
201a

Fig. 18

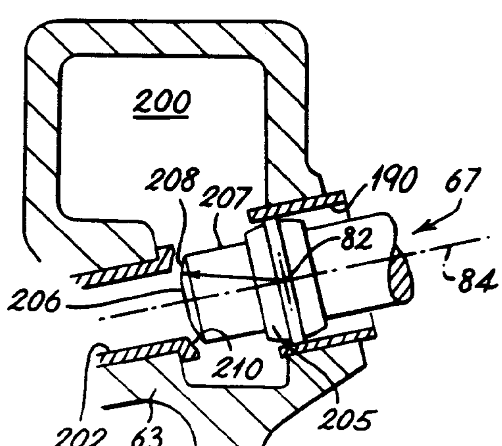
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Fig. 19

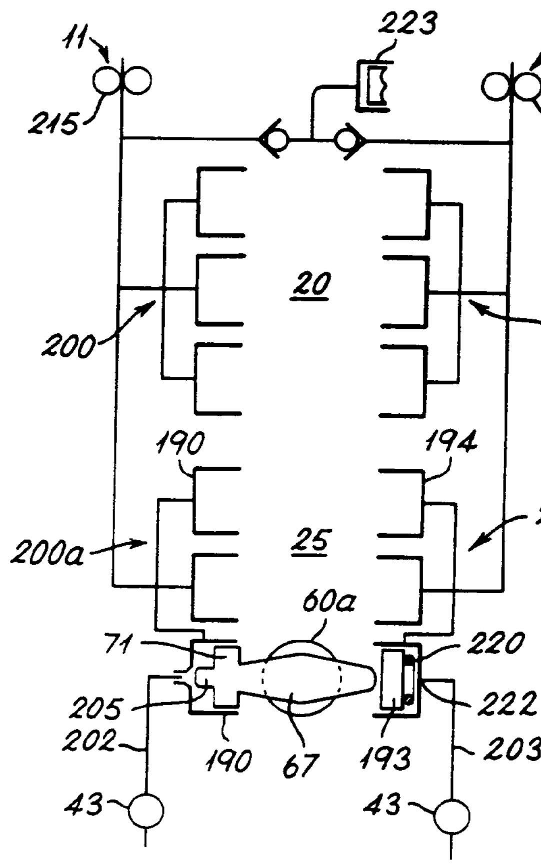
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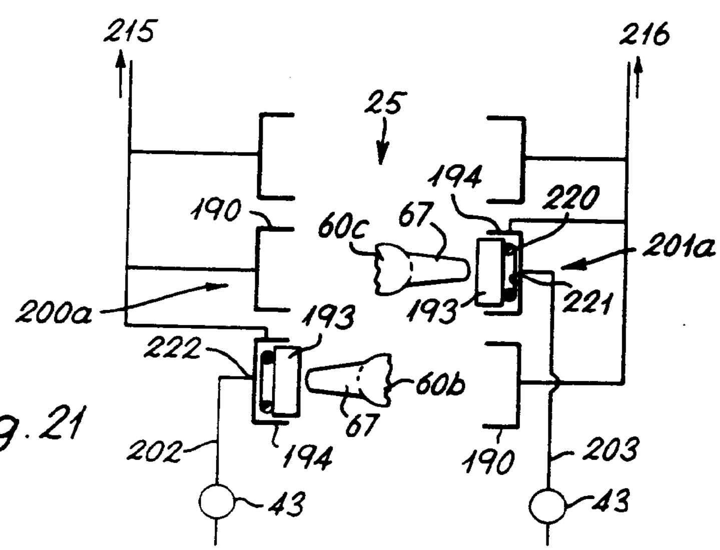
Fig. 20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201a
200a
193
221
193
222
60b
202
194
67
190
203
43
43

Fig. 21